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Analysis of Flow Around a Heated Circular Cylinder

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Abstract: The objective of this study is to investigate the forced convection from and the flow around a heated cylinder. This work presents experimental and computational results for laminar flow around a heated circular cylinder. The experiments were carried out using Particle Image Velocimetry (PIV) in a wind tunnel, and numerical simulations by an in-house code and a commercial software package, FLUENT. This paper presents comparisons for vorticity and temperature contours in the wake of the cylinder.

1. Introduction

The flow around cylindrical bodies has been the subject of intense research for decades, and heat transfer is also of key importance. Here we investigate the effect of heat transfer on the flow from a heated cylinder. Our eventual aim is to design a measuring method for the real-time monitoring of the velocity profile in a channel with a large cross-section (Szabó and Juhász, 2003).

Since the properties of the fluid depend on temperature, and for a heated cylinder the temperature of the fluid varies, it is necessary to consider the most appropriate temperature for calculating the fluid properties and related similarity numbers (Wang and Trávníček, 2001). A typical approach is to use the average of the wall and ambient temperatures (Özisik, 1985; Bejan, 1993).

Here we present experimental and computational results for laminar flow around a heated circular cylinder and compare computed vorticity and temperature contours in the wake of the cylinder. We also discuss which temperature should be used to represent the air properties in the similarity numbers.

2. Measuring set up and results

Experiments were carried out at the Institute of Fluid Dynamics and Thermodynamics, University of Magdeburg, in a Göttingen type wind tunnel with an open section (height 500 mm, width 600 mm, length 1070 mm). A D=10mm diameter and L=600mm length circular cylinder was placed horizontally with its axis perpendicular to the free stream velocity U (see layout in Figure 1). The cylinder was heated electrically, kept at a constant temperature T_w with a potentiometer, and its temperature was measured by a thermocouple.

The flow around the cylinder was measured in a measuring area consisting of a vertical plane perpendicular to the axis of the cylinder. Measurements were carried out using PIV, allowing us to measure for a two-dimensional cross-section of the flow field. Oil fog was added to the flow and the measurement area was illuminated by laser light. The laser acts as a photographic flash for the digital camera, and the particles in the fluid scatter the light. Using this method, we are able to obtain the two-dimensional velocity distribution at different instances. From the velocity distribution streamlines can be determined.

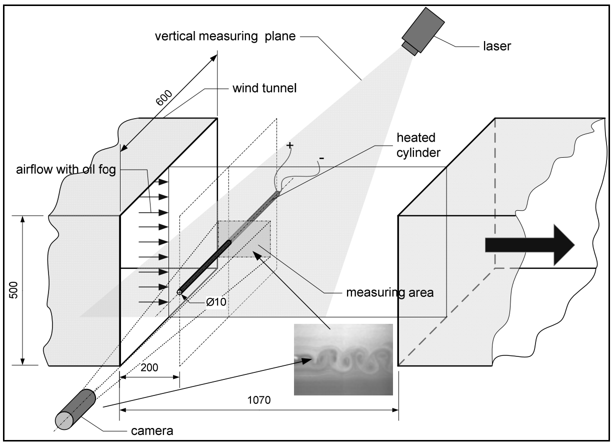

Figure 1. Schematic of experimental setup

From the velocity field **v** we can obtain the vorticity distribution $\varsigma = curl\,\mathbf{v}$ by numerical differentiation. The vorticity distribution shows the precise location, centre, and direction of rotation, allowing us to identify the structure of flow.

Measurements were carried out at the three velocities U=0.3, 0.43, 0.6 m/s, corresponding to $Re_\infty = U\,D / \nu_\infty = 195, 277, 388$, and five cylinder temperature values (T_w = 297, 373, 473, 573, 673 K). Here ν_∞ is the kinematic viscosity of the ambient air (at T_∞). The ambient temperature, and also the temperature of the unheated cylinder, was T_w=T_∞=297K. For the heated cylinder, when calculating similarity numbers, Özisik (1985) and Bejan (1993) suggest basing them on the average or film temperature of the cylinder and the ambient air

$$T_f = \frac{1}{2}\left(T_w + T_\infty\right). \tag{1}$$

Here we define the reference temperature as

$$T_{ref} = T_\infty + c_{ref}\left(T_w - T_\infty\right), \tag{2}$$

which is reduced to equation (1) when $c_{ref} = c_f = 0.5$, where subscript *f* refers to the film value.

The kinematic viscosity belonging to the reference temperature $\nu_{ref} = \nu\left(T_{ref}\right)$ and the related Reynolds number $Re_{ref} = U\,D / \nu_{ref}$ – which takes into account also the cylinder temperature – can also be calculated. The dimensionless vortex shedding frequency or Strouhal number $St = f\ D/U$ for an unheated cylinder is available as a function of Reynolds number (Williamson, 1996; Norberg, 2001).

One set of data consisting of 30 photographs is available for each combination of velocity and temperature. The time-history of the velocity was measured using LDA at a point 70 mm downstream of the center of cylinder and then applied FFT analysis to the signal to determine the vortex shedding frequency.

The results obtained from an analysis of the results can be found in Figure 2, along with computational results and those of Williamson (1996), whose data are for an unheated cylinder at varying free stream velocity *U*.

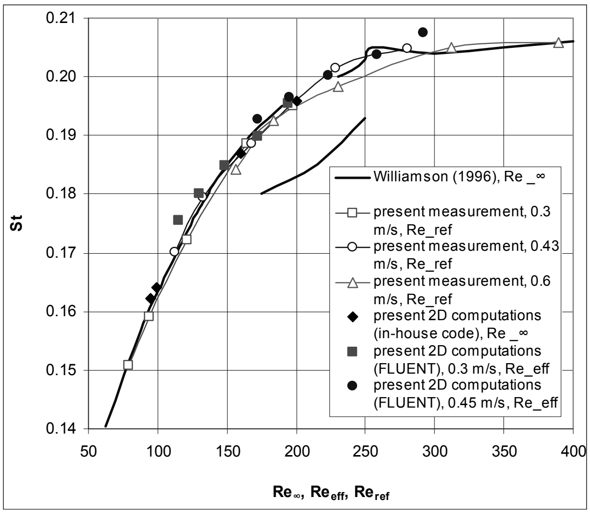

Figure 2. Strouhal number versus Reynolds number for experimental and computational results, compared with Williamson (1996)

The current measurements, however, refer to three constant velocities with the cylinder temperatures varying. We found good agreement with Williamson's results over the domain $Re_{ref} = 79 - 390$ if c_{ref} is defined as a function of temperature ratio $T^* = T_w / T_\infty$ as

$$c_{ref} = 0{,}135\left(T^*\right)^3 - 0.832\left(T^*\right)^2 + 1.626\,T^* - 0.432 \tag{3}$$

shown in Figure 3.

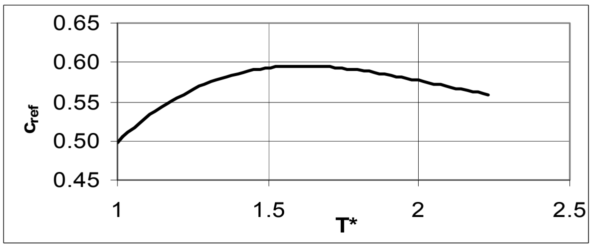

Figure 3. c_{ref} as a function of temperature ratio

Wang et al. (2000), based on their experiments with a cylinder of 1.07 mm in diameter, offer a value of *c*=0.28 independently of the temperature ratio in the domain of $T^* < 2$ and call the related temperature $T_{eff} = T_\infty + 0.28\left(T_w - T_\infty\right)$ effective temperature. The question arises whether the substantial difference between our results and those of Wang et al. is caused by the substantial difference in cylinder diameters. It is worth mentioning, however, that if we replace the 0.28 value of the *c* suggested by Wang et al. (2000) by the c_{ref} defined by equation (3), our test results for Strouhal number differ less than 1% from their values, approximated by $St = 0.2660 - 1.0160 / \sqrt{Re_{eff}}$ for $Re_{eff} < 300$.

One objective of this paper is to determine the Nusselt number. First we checked the criteria for forced convection based on formulae in Wang and Trávniček (2001), and found that all formulae predict forced convection for all of our experimental tests. The heat loss of the cylinder is due to heat convection and radiation, and we neglected the heat conduction, based on Wang and Trávniček's findings that heat conduction is negligible. We measured the current $I\left[A\right]$ and voltage $U_e\left[V\right]$ of the direct current used for heating of the cylinder. We assumed that the electrical power $P_e = U_e\,I$ is fully converted into heat. The cylinder was painted matt black so its emissivity is $\varepsilon \approx 0.97$. The volumetric coefficient of thermal expansion of air is $\beta = 1/(273.15\,K)$, and its heat conduction coefficient k changes with absolute temperature T as $k = 0.0703\,T + 5.5306$. The Stefan-Boltzmann constant is $\sigma = 5.6705 \times 10^{-8}\ W\,m^{-2}\,K^{-4}$. Based on the results of Wang and Trávniček (2001) the Nusselt number belonging to film temperature Nu_f can be written as follows:

$$Nu_f = \frac{1}{\pi\, k_f \left(T_w - T_\infty\right)} \left[\frac{P_e}{L} - \pi D\, \varepsilon\, \sigma \left(T_w^4 - T_\infty^4\right) \right]. \quad (4)$$

Wang and Trávniček (2001) showed that this Nusselt number is a linear function of the square root of the Reynolds number Re_{rep} based on the representative temperature defined by

$$T_{rep} = T_\infty + 0.36\left(T_w - T_\infty\right)$$

over the domain of $40 \le Re_{rep} \le 150$, i.e.,

$$Nu_f = -0.153 + 0.527\, Re_{rep}^{0.5}. \quad (5)$$

We extended the domain of investigation to $Re_{rep} \le 340$. Figure 4 shows our test results together with those of Wang and Trávniček (2001) defined by equation (5). To make our results collapse on the curve, we had to introduce a nominal wall temperature $T_{w,nom}$ and temperature ratio $T_{nom}^* = T_{w,\,nom} / T_\infty$, respectively, and to introduce the correction between temperature ratios T_{nom}^* and $T^* = T_w / T_\infty$ shown in Figure 5. The reason is that the average surface temperature T_w of the heated cylinder for this cylinder size is smaller than the nominal temperature $T_{w,\,nom}$ measured in the interior of the cylinder. The question arises how the average cylinder surface temperature should be measured; this is a future task for investigation.

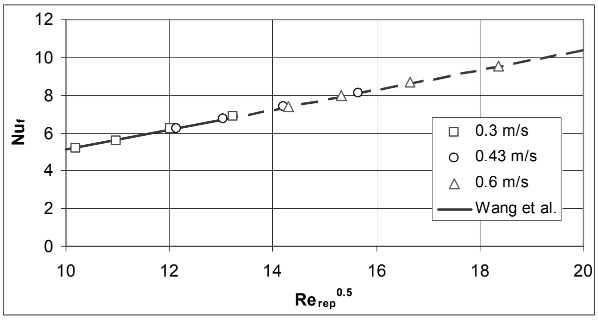

Figure 4. Nusselt number versus Reynolds number for experimental results, compared with Wang and Trávniček (2001)

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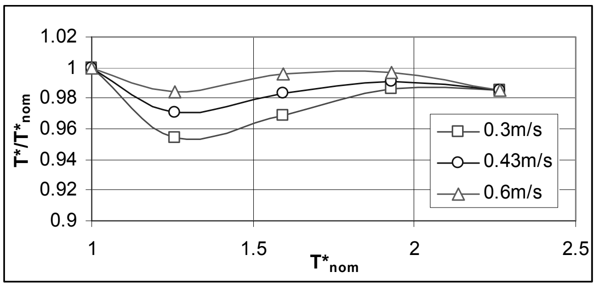

Figure 5. Correction of surface temperature

It is worth mentioning that if we calculate the average Nusselt number from the formula

$$\overline{Nu} = 0.615\left[Re_{eff}\,(T_w / T_\infty)^{0.25}\right]^{0.466}$$

suggested by Hilpert (1933) using Reynolds number Re_f based on the film temperature (1)

$$Nu_f = 0.615\left[Re_f\,(T_w / T_\infty)^{0.25}\right]^{0.466}$$

then our results approximately collapse on the curve suggested by Wang and Trávniček (2001), as can be seen in Figure 6. Computational results can also be seen in Figure 2: filled diamonds show Strouhal numbers obtained by using the in-house code (see section 3) and the filled squares and circles those obtained by FLUENT (section 4).

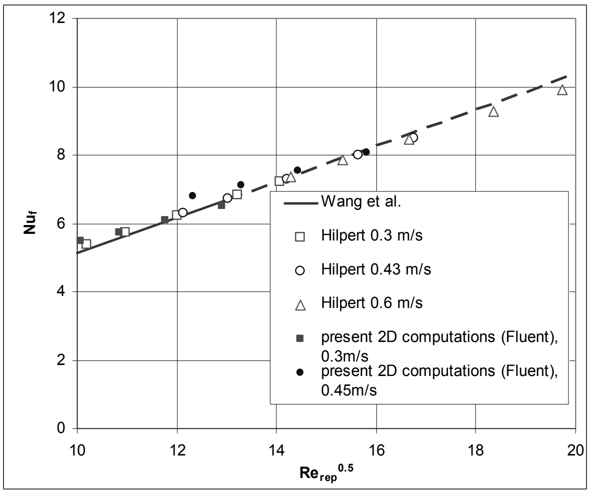

Figure 6. Nusselt number versus Reynolds number for our experimental results evaluated by Hilpert's (1933)' formula, and computational results (FLUENT) compared with those of Wang and Trávniček (2001)

3. Two-dimensional computations with an in-house code

In addition to experiments, numerical simulations were also performed using two different numerical tools. One of them is a 2D in-house code based on a finite difference solution of the governing equations investing low Reynolds number constant property icompressible Newtonian fluid flow around a stationary cylinder. The governing equations are the unsteady Navier-Stokes equations, continuity, a Poisson equation for pressure and an energy equation, all in dimensionless form. The computational domain is characterized by two concentric circles: the inner represents the cylinder surface, the outer the far field. Typical boundary conditions are used for velocity and pressure. The cylinder surface is kept at constant temperature T_w and the ambient temperature T_∞ is also considered to be constant. Forced convection is assumed and thus the buoyancy term is neglected in the Navier-Stokes equations. In this way the flow field is not affected by the temperature but naturally the velocity distribution influences the temperature field. With this method very good agreement was found with the Nusselt number against Reynolds number obtained experimentally (see Baranyi, 2003). The computational method and its validation are described in detail in Baranyi (2003) and Baranyi (2008).

Computations were carried out for both heated and unheated stationary cylinder. The obtained Strouhal numbers can be seen as filled diamonds in Figure 2. Figure 7 shows vorticity contours belonging to the same instant at Re_∞=200 obtained from experimental data (top), computational results by in-house code (middle) and FLUENT (bottom). The computational results compare well with experimental results obtained by PIV.

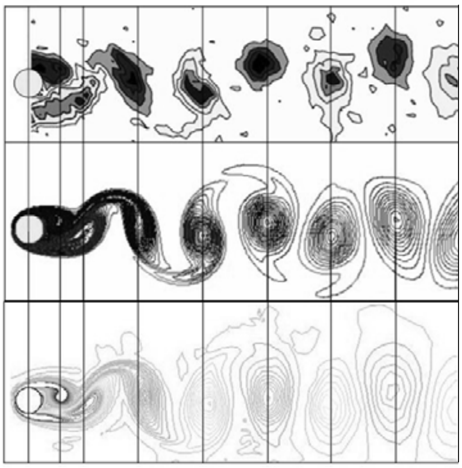

Figure 7. Vorticity contours obtained experimentally (top) and numerically (middle: in-house code, bottom: FLUENT) for unheated cylinder at Re_∞=200

4. Numerical simulation with commercial software

For the numerical solution of the governing equations in two-dimensions the commercial software package FLUENT V6.3.26 is employed, which uses the finite volume method. The top and bottom boundary is modeled as symmetric, the inlet velocity are U=0.3 and 0.45 m/s, and ambient temperature is constant (T_∞=297 K).The effect of gravity is neglected. The cylinder surface is kept at different constant temperatures of T_w=297, 373, 473, 573, 673 K. The computational domain is characterized by two concentric circles: the inner represents the cylinder surface, the outer the far field. The numerical grid consists of 28800 elements. The fluid properties are not constant; the effect of temperature is taken into account.

As mentioned earlier, computational results for Strouhal number are shown and compare reasonably well with the results of Williamson (1996) for unheated cylinders, if Re is computed based on effective temperature as suggested by Wang et al. (2000), shown in Figure 2. Figure 6 shows the calculated film Nusselt number Nu_f using Hilpert formula versus the square-root of Re_{rep}. Results compare reasonably well with results on the straight line suggested by Wang and Trávniček (2001). Finally, the bottom figure in Figure 7 shows vorticity contours obtained by FLUENT, and again compare well with experimental and other computational data.

5. Conclusions

Our experimental and computational results for heated and unheated cylinders in terms of *St-Re* relationship agree reasonably well (by introducing effective and representative temperatures) with those of Williamson (1996) obtained for unheated cylinder. Reasonable agreement was found in the *Nu-Re* relationship with Wang and Trávniček (2001). The discrepancies found between our and other authors' results, however, require further investigations.

6. Acknowledgements

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